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LAWRENCE RADIATION LABORATORY

An Ultra-High Vacuum Flange Seal

While developing a new ultra-high vacuum seal needed for various pipes and pieces of equipment in 1960, Tom Batzer, an engineer on Project Sherwood at Lawrence Radiation Laboratory in Livermore, California had to determine the size and spacing of the flange bolts and their torque.

Project Sherwood concerns controlled fusion research. Vacuums of less than 10^{-10} Torr* are needed and systems must be baked out by heating to 400°C or more to drive off adsorbed gases. Bulk diffusion of gas from metal also occurs during bake-out, but stops for all practical purposes after the system is returned to lower temperatures. Metal seals are used because elastomers cannot withstand the heating and also because evaporation of molecules from rubber gaskets could degrade the vacuum. Part of a typical vacuum system showing flanges of many different sizes, all of which must be sealed, is shown in Exhibit 1.

Tom Batzer outlined the evolution of all-metal vacuum seals: "For years metal gaskets were just massive flat flanges with rings of copper wire, the ends butt welded together, clamped between them. But the copper work hardens and dents the flanges and it creeps during bake-out so that bolt tension can relax and the seal is lost. The copper also oxidizes at high temperatures and the low density oxide forms a leak path."

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* One Torr equals one mm of mercury.

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Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by John A. Alic under the direction of Karl H. Vesper with financial support from the National Science Foundation.

"The next obvious step was to try a material that won't oxidize -- gold wire can be used with flat flanges, but it still creeps and work hardens."

Figure 1 of Exhibit 2 shows a refinement -- gold wire 10 to 20 mils in diameter is clamped between asymetric flanges so that creep is limited. Tom said, "These require very close tolerances which limit their diameter to about 8 inches and they're also hard to handle."

In 1956 Westinghouse Research Laboratories in Pittsburgh designed a copper "pinch gasket" while doing some work for Lawrence. This gasket is sketched in Figure 2 of Exhibit 2; the copper ring is about 50 mils thick. Tom pointed out that creep and oxidation are still problems with this design, but in 1957 Lawrence modified it to the configuration shown in Figure 3 of Exhibit 2. Instead of just being pinched, the copper is coined (trapped and compressed) between the flanges. The coining of the copper keeps air from reaching the seal area, preventing oxidation, and the compression of the copper stores energy so that bolt tension is not lost when the copper creeps. Tom said, "Energy storage is also needed to retain a good seal when differential expansion occurs during thermal cycling. Unfortunately this seal requires very close tolerances, is easy to nick and hard to repair, and is still asymetric."

Tom continued, "In 1960, during a meeting, one of the physicists said, 'Why doesn't someone design a simple, reliable flange seal?' This got us back to work and we started testing all the commercial seals that were available. None of them worked, but one day I was watching a vendor working on a gasket with a cross-section shaped like the letter 'K' that they'd been trying to make seal for weeks. I finally said to him, 'I could put some aluminum foil in there and it would work better.' So then I had to try it -- and it did work."

This first set-up is shown in Figure 4 of Exhibit 2. Tom said, "Then we had to sit down and analyze the foil seal. We provided for energy storage by allowing flange rotation. The seal face is machined at an angle slightly greater than the angle through which the flanges rotate at full bolt load. There is contact across only about half the seal width. Our first try, with flat seal faces, would never have been successful except the seal the vendor was testing had a bolt circle of only 3 inches, so flange rotation wasn't too important. We use 1-1/2 mil dead soft foil, which yields and flows into the seal asperities when the bolt load reaches 600 to 700 lbs per lineal inch of seal."

Tom's final seal design, upon which he has a patent (assigned to the AEC) is shown in Exhibit 3 for a four inch tube. The foil seals have been used in diameters ranging from 1/2 inch to 48 inches and can also be used with non-circular flanges. Lawrence uses flanges and bolts of type 304 stainless steel. A 32 microinch finish with concentric tool marks is specified to limit creeping and extrusion of the foil outwards. The foil seals are almost as cheap as rubber O-rings in small quantities and are actually less expensive when produced on automatic machinery.

After finding that a foil seal would work, Tom had several design parameters to determine. Among these were bolt size, torque and spacing. He thought that these could be found by going through the three steps below:

1. Determination of the total force to be applied to the flanges through the bolts. Tom said, "We wanted to store the maximum possible amount of energy in flange rotation, so we go past the yield point of the foil, to a load which just about yields the extreme fibers of the flange.* The question was how to do this."
2. Finding the bolt spacing. Tom said, "It appeared to me that once the flange load is fixed, bolt spacing is limited by flange bowing between the bolts. With 1-1/2 mil foil we wanted to limit the maximum deflection to a few tenths of a mil."
3. Determination of a bolt size and torque to give the flange load found in (1) with the spacing found in (2). Tom said, "We lubricate the bolts with a colloidal dispersion of molybdenum disulfide to increase the tension for a given torque and to prevent galling. Unlubricated bolts develop only about a twelfth of their theoretical tension, while with careful lubrication we get 35 to 45% of the theoretical force."

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* The yield point of annealed 304 stainless steel is 35,000 psi. Cold worked, the yield point is 85,000 psi.

Exhibit 1: Vacuum Pumping System.

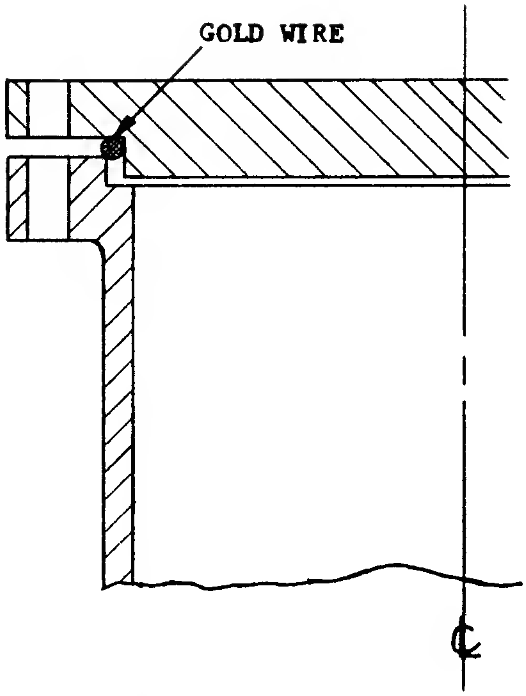


FIGURE 1

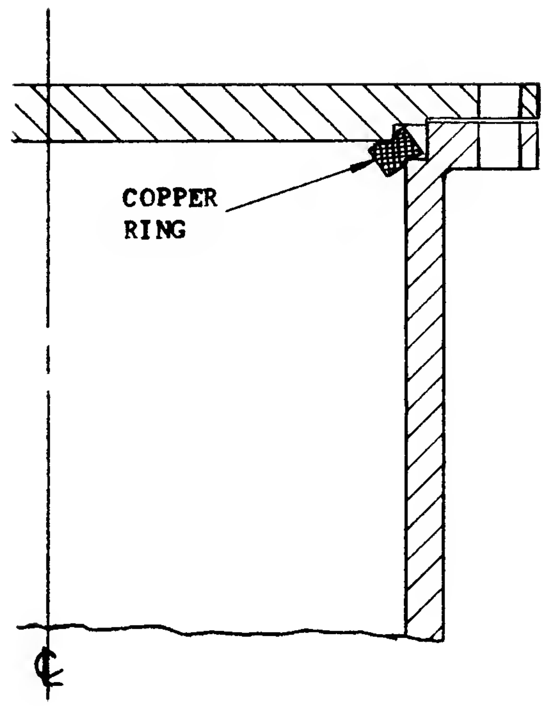


FIGURE 2

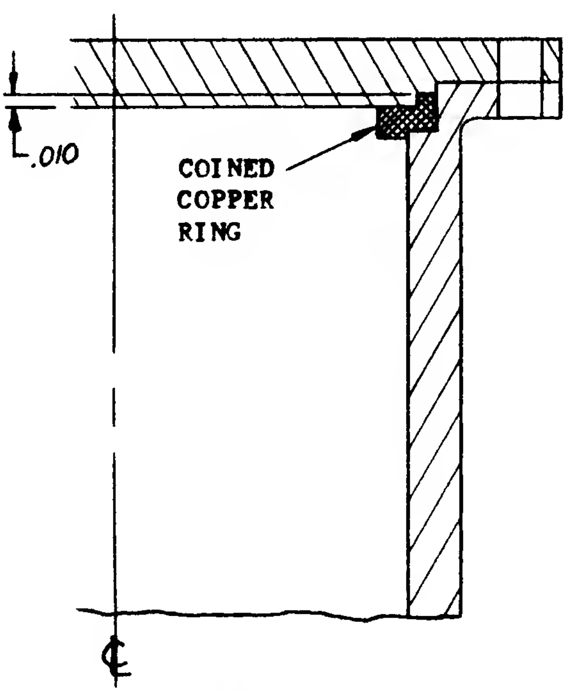


FIGURE 3

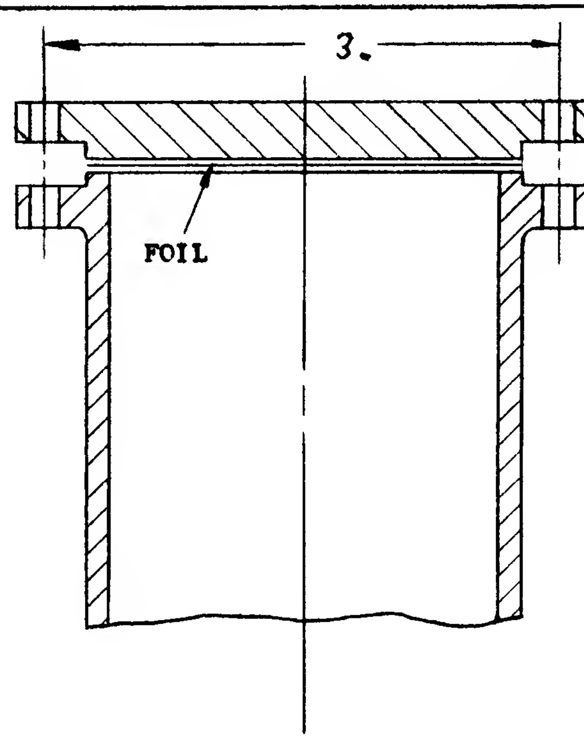
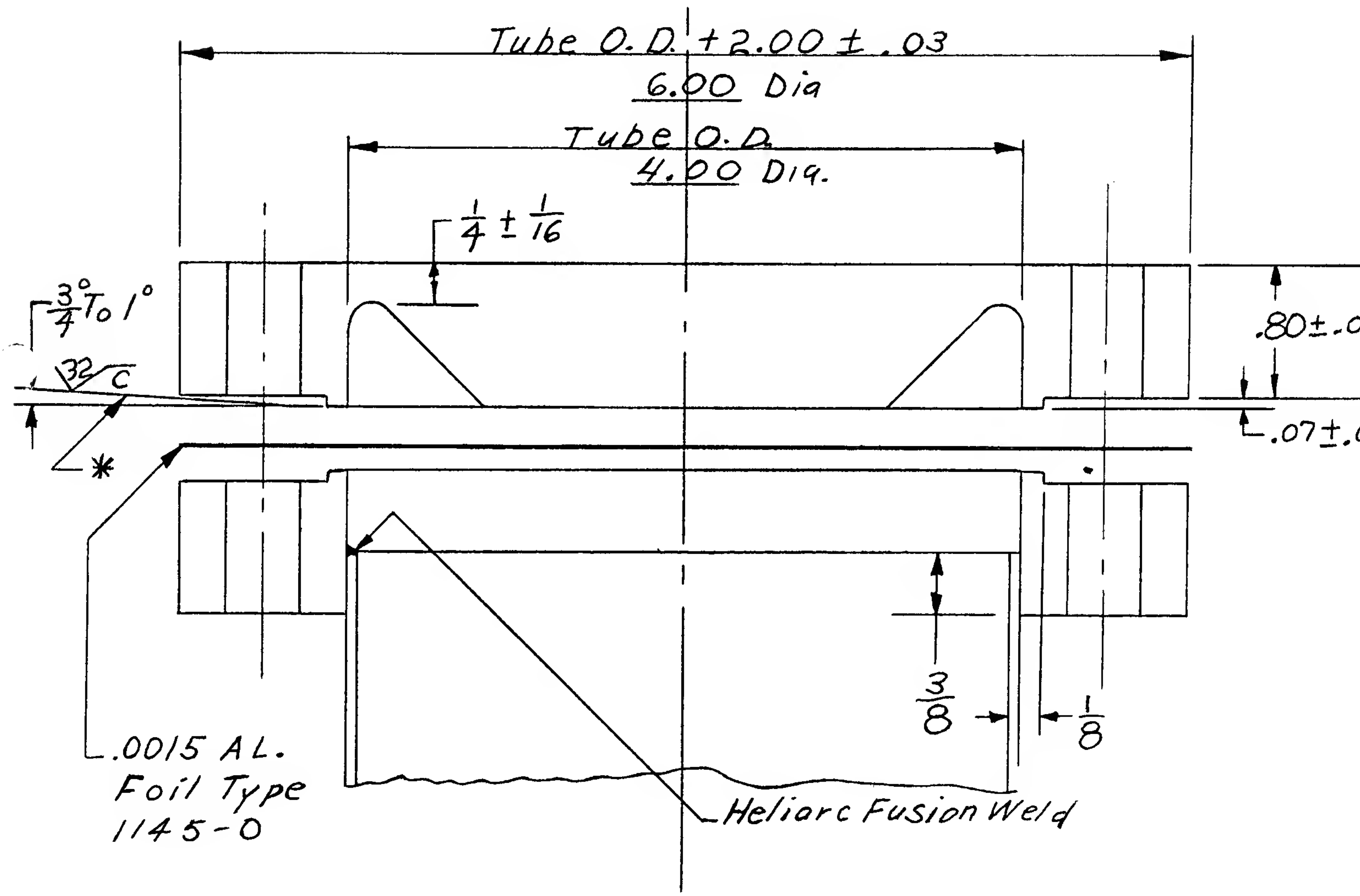


FIGURE 4

<u>Exhibit 2</u>: Flange Seals.

Addendum 1, T. H. Batzer, Jan. 19, 1963

Four Inch Tube Size



*Gasket surface warpage not to exceed .001 T.I.R.

Exhibit 3: Page from a Lawrence ENGINEERING NOTE Detailing a Four Inch Flange.

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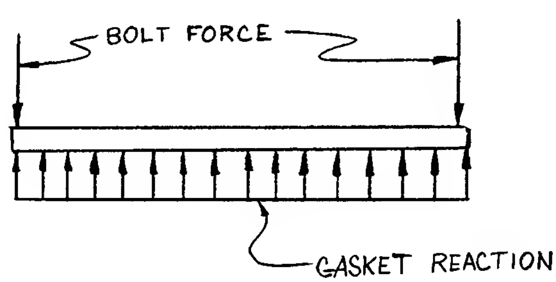
Conclusion

Tom Batzer found the stress in the flanges by using the formula $s_{max} = MR/(I_o/c)$ where M = moment due to bolt force and gasket reaction, R = radius to centroid of flange cross-section, I_o= moment of inertia of cross-section and c = distance from neutral axis to extreme fiber. The equation is from Roark, Formulas for Stress and Strain, Third Edition, page 230. His calculations showed that he could use about 2000 to 3000 lbs. per lineal inch of seal to get to near the yield point. He then calculated flange bowing by treating the section of a flange between two bolts as a straight beam loaded on one side by the bolts and on the other by a uniformly distributed gasket reaction as shown below:



This gave him a bolt spacing of 2 inches and he chose 3/8 inch cold headed and cold rolled bolts torqued to 300 to 350 in-lbs to give the desired flange load. Tom said, "This stresses the bolts to just about their yield point -- I ran a little statistical analysis and broke a lot of bolts to check it. It's okay to do this since stainless is fairly ductile. Quarter-inch bolts wouldn't have been strong enough and half-inch bolts would have required too much torque to develop the force we wanted."